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Performance evaluation of medium grade low heat rejection diesel engine with carbureted methanol and crude jatropha oil

V.S. Murali Krishna M [a], V.R. Seshagiri Rao V [a], Kishen Kumar Reddy T [b], V.K. Murthy P [c,*]

[a] Mechanical Engineering Department, Chaitanya Bharathi Institute of Technology, Gandipet, Hyderabad 500075, Andhra Pradesh, India
[b] Mechanical Engineering Department, J.N.T.U. College of Engineering, Kukatpally, Hyderabad 500085, Andhra Pradesh, India
[c] Jaya Prakash Narayan Educational Society Group of Institutions, Mahabubnagar 509001, Andhra Pradesh, India

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ABSTRACT

Vegetable oils and alcohols (ethanol and methanol) are important substitutes for diesel fuel as they are renewable in nature. However drawbacks associated with vegetable oils (high viscosity and low volatility) and alcohols (low cetane number) call for engine with hot combustion chamber with its significant characteristics of higher operating temperature, maximum heat release, higher brake thermal efficiency (BTE) and ability to handle the lower calorific value fuel. Methanol was inducted into the engine through a variable jet carburetor, installed at the inlet manifold of the engine at different percentages of crude vegetable oil at full load operation on mass basis. Crude vegetable oil was injected at near end of compression stroke. Performance was evaluated with engine with LHR combustion chamber consisting of air gap (3mm) insulated piston with superni (an alloy of nickel) crown and air gap insulated liner with superni insert with mixture of carbureted methanol and crude vegetable oil with varied injector opening pressure and injection timing. Comparative studies were made with crude vegetable oil operation on engine with LHR combustion chamber at similar operating conditions. Performance parameters and exhaust emissions were determined at various values of brake mean effective pressure. Aldehydes were measured by the 2,4, dinitrophenyl hydrazine (DNPH) method. Combustion characteristics were measured with top dead center (TDC) encoder, pressure transducer, console and special pressure–crank angle software package at full load operation of the engine.

The optimum injection timing with crude vegetable oil operation on LHR combustion chamber was 29° bTDC. The maximum induction of methanol was 55% at recommended injection timing (27° bTDC), while it was 50% at optimum injection timing. With maximum induction of methanol, at an injector opening pressure of 190 bar, at recommended injection timing, engine with LHR combustion chamber increased peak brake thermal efficiency by 6%; at full load operation it decreased brake specific energy consumption by 2%, exhaust gas temperature by 16%, coolant load by 11%, volumetric efficiency by 6%, sound levels by 8%, particulate matter by 45%, NO_x emissions by 46%, increased formaldehyde emissions and acetaldehyde emissions drastically, increased peak pressure by 23% and maximum rate of pressure rise by 17% when compared with crude jatropha oil operation at similar operating conditions.

Contents

* Corresponding author. Tel.: +91 9490116544; fax: +91 8542276144.
E-mail addresses: mvsmk@cbit.ac.in (V.S. Murali Krishna M), krishnamurthy_venkata@yahoo.co.in (V.K. Murthy P).

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1. Introduction

In the context of depletion of fossil fuels, ever increase of fuel prices in International Market causing economic burden on developing countries, increase of pollution levels with fossil fuels, the search for alternative fuels has become pertinent. It has been found that vegetable oils are promising substitute for diesel fuel, because their properties are comparable to those of diesel fuel. They are renewable and can be easily produced.

Rudolph Diesel, the inventor of the diesel engine that bears his name, experimented with fuels ranging from powdered coal to peanut oil [1]. Several researchers experimented the use of vegetable oils as fuel on CE and reported that the performance was poor, citing the problems of high viscosity, low volatility and their polyunsaturated character causing piston ring sticking, injector and combustion chamber deposits, fuel system deposits, reduced power, reduced fuel economy and increased exhaust emissions [2–10].

On the other hand alcohols are renewable and volatile fuels. There are many methods of inducting alcohols in diesel engines, out of which blending and carburetion techniques are simple. Investigations were carried out with blended alcohol with diesel fuel in conventional engine and reported performance improved with blends [11–15]. Exhaust emissions of particulate matter and nitrogen oxides (NO_x) decreased in comparison with pure diesel operation on conventional engine.

However, methanol has a low cetane number (less than 10). Hence engine modification is necessary if carbureted methanol is used as fuel in diesel engine. The drawbacks associated with the crude vegetable oil and methanol as fuels in diesel engine call for hot combustion chamber provided by LHR combustion chamber.

The major concept of engine with LHR combustion chamber is to reduce heat loss to the coolant, by providing thermal insulation in the path of heat flow to the coolant. Engines with LHR combustion chamber are classified depending on the degree of insulation, such as ceramic coated combustion chambers (low grade), air gap insulated combustion chambers (medium grade) and combination of these two (high grade).

Experiments were conducted on engine with medium grade LHR combustion chamber with vegetable oils with varied injector opening pressure and injection timing [16–18]. It was reported from their investigations that medium grade LHR combustion chamber improved the performance in comparison with pure diesel operation on conventional engine. However, it drastically increased NO_x emissions. Studies were also made with carbureted alcohols (ethanol/methanol) on medium grade LHR combustion chamber with diesel operation [19–21]. Alcohol (ethanol/methanol) was inducted into the engine by means of variable jet carburetor at different percentages of diesel fuel at full load operation and diesel was injected in conventional manner at the end of compression stroke. Exhaust emissions of particulate matter and NO_x levels decreased with the carburetion technique. The advantage of this method over the previous method was that more amount of alcohol (ethanol/methanol) can be inducted into the engine. It was reported that optimum induction of alcohol (ethanol/methanol) was 35% with conventional engine and 50% with medium grade LHR combustion chamber with diesel operation on mass basis.

Carbureted alcohols were used in engine with high grade LHR combustion chamber with vegetable oils and reported that performance improved with LHR combustion chamber [22–24].

In order to take advantage from high cetane number and high volatility, both vegetable oils and alcohols have to be used in engine with LHR combustion chamber.

The present paper attempted to evaluate the performance of the engine with medium grade LHR combustion chamber, which consisted of air gap insulated piston and air gap insulated liner with crude jatropha oil (CJO), with carbureted methanol, with varied injector opening pressure and injection timing and compared with crude jatropha oil at the similar operating conditions.

2. Material and method

Fig. 1 shows the assembly details of air gap insulated piston and air gap insulated liner. Engine with LHR combustion chamber contained a two-part piston; the top crown made of low thermal conductivity material, superni-90 screwed to aluminum body of the piston, providing a 3 mm-air gap in between the crown and the body of the piston. The optimum thickness of air gap in the air gap piston was found to be 3 mm for improved performance of the engine with superni insert with diesel as fuel [16]. A superni-90 insert was screwed to the top portion of the liner in such a manner that an air gap of 3 mm was maintained between the insert and the liner body. At 500 °C the thermal conductivities of superni-90 and air are 20.92 and 0.057 W/m K respectively.

The schematic diagram of the experimental setup used for the investigations on the engine with LHR combustion chamber with jatropha oil and carbureted methanol is shown in Fig. 2. The specifications of the experimental engine are given in Table 1.

The combustion chamber consisted of a direct injection type with no special arrangement for swirling motion of air. The engine was connected to an electric dynamometer for measuring its brake power. Methanol was inducted through the variable carburetor jet, located at the inlet manifold of the engine at different percentages of vegetable oil by mass basis and crude vegetable oil (CJO) was injected in the conventional manner. Two separate fuel tanks and buret arrangements were made for measuring crude jatropha oil and methanol consumptions. Air-consumption of the engine was

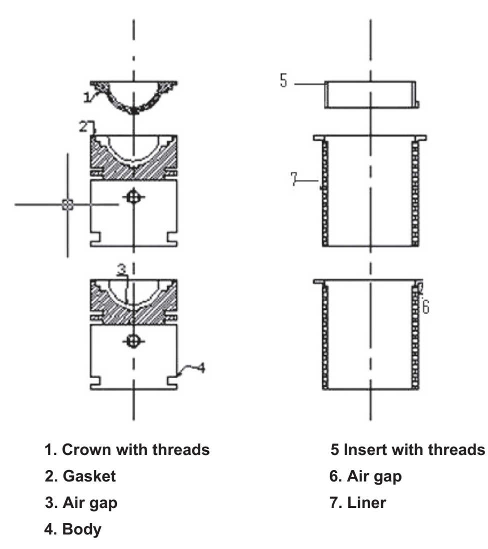


Fig. 1. Assembly details of air gap insulated piston and air gap insulated liner.

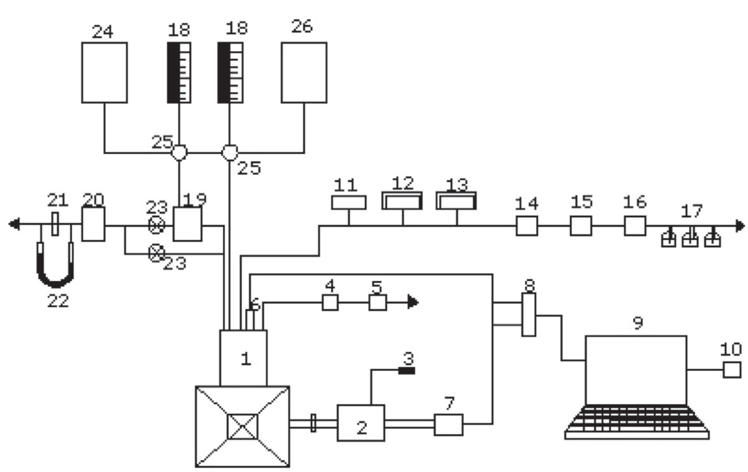
Nomenclature

ρ_a	density of air, kg/m^3
ρ_d	density of fuel, gm/cc
η_d	efficiency of dynamometer, 0.85
a	area of the orifice flow meter in square meter, $(\pi \times d^2)/4$
BP	brake power of the engine, kW
C	number of carbon atoms in fuel composition
C_d	coefficient of discharge, 0.65
Cp	specific heat of water in kJ/kg K
D	bore of the cylinder, 80 mm
d	diameter of the orifice flow meter, 20 mm
DF	diesel fuel
H	number of hydrogen atoms in fuel
HSU	Hartridge smoke unit
I	ammeter reading, ampere
h	difference of water level in U-tube water manometer in cm of water column
IT	injection timing, degree bTDC
k	number of cylinders, 01
L	stroke of the engine, 110 mm
m_a	mass of air inducted in engine, kg/h
m_f	mass of fuel in kg/h
m_w	mass flow rate of coolant (water), kg/s
n	power cycles per minute, N/2,
N	speed of the engine, 1500 rpm
P_a	atmosphere pressure in mm of mercury
R	gas constant for air, 287 J/kg K
t	time taken for collecting 10 cc of fuel, second
T_a	room temperature, degree centigrade
T_I	inlet temperature of water, degree centigrade
T_o	outlet temperature of water, degree centigrade
V	voltmeter reading, Volts
V_s	stroke volume, m^3

measured by air-box (orifice flow meter, U-tube water manometer). The naturally aspirated engine was provided with water-cooling system in which inlet temperature of water was maintained at 80 °C by adjusting the water flow rate. The engine oil was provided with a pressure feed system. No temperature control was incorporated, for measuring the lube oil temperature. Copper shims of suitable size were provided in between the pump body and the engine frame, to vary the injection timing and its effect on the performance of the engine was studied, along with the change of injection pressures from 190 bar to 270 bar (in steps of 40 bar) using nozzle testing device. The maximum injector opening pressure was restricted to 270 bar due to practical difficulties involved. Exhaust gas temperature (EGT) was measured with thermocouples made of iron and iron-constantan. The accuracy of exhaust gas indicator was 1%. The speed of the engine was measured with tachometer with accuracy 1%.

Exhaust emissions of smoke and NO_x were recorded by AVL smoke meter and Netel Chromatograph NO_x analyzer respectively at various values of brake mean effective pressure. The specifications of the analyzers are given in Table 2.

With alcohol–vegetable mixture operation, the major pollutant emitted from the engine is aldehydes. These aldehydes are carcinogenic in nature, which are harmful to human beings. The measure of the aldehydes is not sufficiently reported in the literature. The DNPH method was employed for measuring aldehydes in the experiment [21]. The exhaust of engine was purified by means of filter and the measured quantity (2 l/m) was sent through rotometer. The exhaust of the engine was heated up to 140 °C by means of a heater provided in the circuit. The exhaust of the engine was bubbled through dinitrophenyl hydrazine (2,4 DNPH) solution. The hydrazones formed were extracted into chloroform and were analyzed by employing high performance liquid chromatography (HPLC) to find the percentage concentration of formaldehyde and acetaldehyde in the exhaust of the engine. The advantage of this method was determination of both formaldehyde concentration and acetaldehyde concentration simultaneously in the exhaust of the engine.



1.Engine, 2.Electical Dynamometer, 3.Load Box, 4. Outlet jacket water temperature indicator, 5.Outlet-jacket water flow meter Orifice meter, 6. Piezo-electric pressure transducer, 7. TDC encoder 8.Console, 9. Pentium Personal Computer, 10. Printer, 11.Exhaust gas temperature indicator, 12.AVL Smoke meter, 13. Netel Chromatograph NOx Analyzer, 14. Filter, 15.Rotometer, 16.Hetaer,17. Round bottom flask containing DNPH solution, 18.Burette, 19. Variable jet carburetor, 20. Air box, 21.Orifice meter, 22. U-tube water manometer, 23.Bypass valve, 24.Methanol tank, 25. Three-way valve, 26 Vegetable oil tank.

Fig. 2. Experimental set-up.

Table 1
Specifications of the test engine.

Description	Specification
Engine make and model	Kirloskar (India) AV1
Maximum power output at a speed of 1500 rpm	3.68 kW
Number of cylinders × cylinder position × stroke	One × vertical position × four-stroke
Bore × stroke	80 mm × 110 mm
Method of cooling	Water cooled
Rated speed (constant)	1500 rpm
Fuel injection system	In-line and direct injection
Compression ratio	16:1
BMEP @ 1500 rpm	5.31 bar
Manufacturer's recommended injection timing and pressure	27° bTDC × 190 bar
Dynamometer	Electrical dynamometer
Number of holes of injector and size	Three × 0.25 mm
Type of combustion chamber	Direct injection type
Fuel injection nozzle	Make: MICO-BOSCH No.: 0431-202-120/HB
Fuel injection pump	Make: BOSCH: NO.: 8085587/1

Table 2
Specifications of the analyzers.

Name of the analyzer	Measuring range	Precision	Resolution
Sound analyzer	0–200 dB	1 dB	1 dB
AVL particulate matter analyzer	0–100 HSU	1 HSU	1 HSU
Netel chromatograph NO_x analyzer	0–2000 ppm	2 ppm	1 ppm

Table 3
Properties of test fuels.

Test fuel	Viscosity at 25 °C (centipoise)	Density at 25 °C	Cetane number	Low calorific value (kJ/kg)
Diesel	12.5	0.84	55	42,000
Crude jatropha oil (CJO)	125	0.90	45	36,000
Methanol	–	0.81	03	19,740

Piezo electric transducer, fitted on the cylinder head to measure pressure in the combustion chamber, was connected to a console, which in turn was connected to a Pentium personal computer. TDC encoder provided at the extended shaft of the dynamometer was connected to the console to measure the crank angle of the engine. A special P–θ software package evaluated the combustion characteristics such as peak pressure (PP), time of occurrence of peak pressure (TOPP) and maximum rate of pressure rise (MRPR) from the signals of pressure and crank angle at the full load operation of the engine. The accuracy of the instrumentation was 1%.

The whole seeds from the plant '*Jatropha curcas*' can be crushed to yield about 25% crude jatropha oil. Double crushing can increase the yield to 28.5% and solvent extraction to 30%. The properties of the diesel, vegetable oil and methanol used in this work are presented in Table 3.

Test fuels used in the experiment were pure diesel, pure crude vegetable oil, carbureted ethanol/methanol along with injected crude vegetable oil. Different injector opening pressures attempted in this experiment were 190 bar, 230 bar and 270 bar. Various injection timings attempted in the investigations were 27–34° bTDC.

Definitions of used values:

$$m_f = \frac{10 \times \rho_d \times 3600}{t \times 1000}$$

$$\mathrm{BP} = \frac{\mathrm{VI}}{\eta_d \times 1000}$$

$$\mathrm{BTE} = \frac{\mathrm{BP} \times 3600}{m_f \mathrm{CV}} \text{ for diesel/vegetable oil operation}$$

$$\mathrm{BTE} = \frac{\mathrm{BP} \times 3600}{m_f \mathrm{CV} + m_{\mathrm{al}} \mathrm{CV}_{\mathrm{al}}} \text{ for alcohol and vegetable oil operation}$$

m_{al} = Mass of alcohol in kg/h, $\mathrm{CV}_{\mathrm{al}}$ = Calorific value of alcohol

$$\mathrm{BSEC} = \frac{1}{\mathrm{BTE}}$$

$$\mathrm{BP} = \frac{BMEP \times 10^5 \times LAnk}{60{,}000}$$

$$\mathrm{CL} = m_w c_p (T_o - T_i)$$

$$m_a = C_d a \sqrt{2 \times 10 \times gh\rho_a} \times 3600$$

$$\eta_v = \frac{m_a \times 2}{60 \times \rho_a N V_s}$$

$$\rho_a = \frac{P_a \times 10^5}{750 \times RT_a}$$

% of alcohol inducted

$$= \frac{\text{mass of alcohol}}{\text{mass of alcohol} + \text{mass of vegetable oil at full load operation}}$$

Optimum injection timing: It is the injection timing at which maximum thermal efficiency of the engine is obtained at all loads.

3. Results and discussion

3.1. Performance parameters

The optimum injection timing with crude vegetable oil with conventional engine was 32° bTDC, while it was 30° bTDC with engine with LHR combustion chamber.

The optimum injection timing with diesel with conventional engine was 31° bTDC, while it was 29° bTDC with engine with LHR combustion chamber. As diesel has a high cetane number, the optimum injection timing was obtained earlier with diesel operation on both versions of the engine.

Investigations were carried out with the objective of determining the factors that would allow maximum use of alcohol in diesel engine with best possible efficiency at all loads. Fig. 3 indicates that BTE increased up to 80% of the full load and beyond that load it decreased for all the test fuels. This was due to the increase in fuel efficiency up to 80% of the full load and beyond that load it was because of reduction of oxygen–fuel ratio and volumetric efficiency.

Conventional engine with crude jatropha oil showed the deterioration in the performance at all loads when compared to the pure diesel operation. This was due to low calorific value, high viscosity of the vegetable oil. The amount of air entrained by the fuel spray was reduced, since the fuel spray plume angle was reduced, resulting in slower fuel–air mixing [10]. In addition, less air entrainment by the fuel spray suggested that the fuel spray penetration might increase and result in more fuel reaching the combustion chamber walls [9]. Furthermore droplet mean

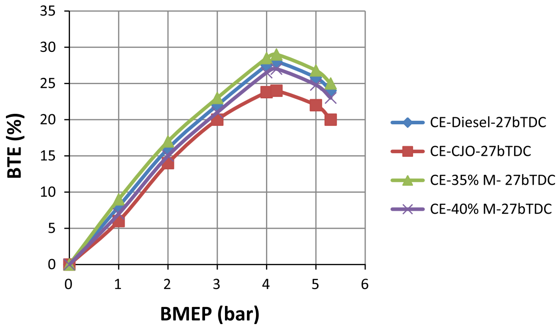


Fig. 3. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in conventional engine (CE) at different percentages of methanol (M) induction at 27° bTDC at an injector opening pressure of 190 bar.

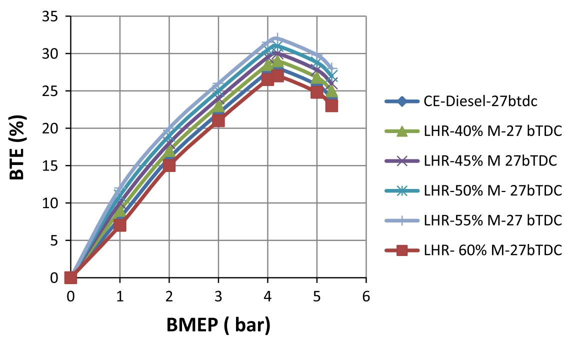


Fig. 4. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) in engine with LHR combustion chamber at different percentages of methanol (M) induction at 27° bTDC at an injector opening pressure of 190 bar.

diameters (expressed as Sauter mean) were larger for vegetable oil leading to the reduction of the rate of heat release as compared to diesel fuel [9]. This also, contributed to the higher ignition (chemical) delay of the crude vegetable oil due to lower cetane number. According to the qualitative image of the combustion under the crude jatropha oil operation with conventional engine, the lower brake thermal efficiency was attributed to the relatively retarded and lower heat release rates.

Brake thermal efficiency increased with methanol induction. It increased at all loads with 35% methanol induction. This was due to improved homogeneity of the mixture with the presence of methanol, decreased dissociated losses, specific heat losses and cooling losses due to lower combustion temperatures. This was also due to high heat of evaporation of methanol, which caused the reduction in the gas temperatures resulting in a lower ratio of specific heats leading to more efficient conversion of heat into work. Induction of methanol resulted in more moles of working gas, which caused high pressures in the cylinder. The observed increase in the ignition delay period would allow more time for fuel to vaporize before the ignition started. This means that higher burning rates resulted in more heat release rate at constant volume, which was a more efficient conversion process of heat into work. Brake thermal efficiency decreased at all loads with methanol induction more than 35%. This was due to increase of ignition delay and reduction of combustion temperatures.

Curves in Fig. 4 indicate that engine with LHR combustion chamber showed an improvement in the performance with the carbureted methanol at all loads when compared to the pure diesel operation on conventional engine. This was due to recovery of heat from the hot insulated components of LHR combustion chamber due to high latent heat of evaporation of the methanol, which lead to increase in thermal efficiency. The maximum induction of methanol was 55% in engine with LHR combustion chamber, which showed improvement in the performance at all loads when compared to pure diesel operation on conventional engine. However when the methanol induction was increased more than 55% in engine with LHR combustion chamber, brake thermal efficiency decreased at all loads when compared with pure diesel operation on conventional engine. This was due to the increase of ignition delay.

The maximum induction of methanol was limited to 50% in the engine with LHR combustion chamber at optimum injection timing (30° bTDC) against 55% induction at 27° bDC with vegetable oil operation. The maximum induction of methanol remained the same in conventional engine at its optimum injection timing (32° bTDC) as in the case of 27° bTDC with vegetable oil operation.

The maximum induction of methanol was limited to 45% in the engine with LHR combustion chamber at optimum injection timing (29° bTDC) against 50% induction at 27° bTDC with diesel operation [20]. The maximum induction of methanol remained the same in conventional engine at its optimum injection timing (31° bTDC) as in the case of 27° bTDC with diesel operation [20].

From Fig. 5, it is noticed that engine with LHR combustion chamber with 50% methanol induction at its optimum injection timing showed improved performance at all loads when compared with other versions of the combustion chamber. This was due to higher amount of methanol substitution and improved combustion at advanced injection timing that caused better evaporation leading to produce higher brake thermal efficiency.

There is a limitation to use methanol due to a low cetane number and having higher self-ignition temperature than vegetable oil to use in conventional engine without increasing injector opening pressure because as percentage of methanol increased, more heat was utilized to evaporate methanol and less heat was available to evaporate vegetable oil. Therefore a major quantity of alcohol which burned late in the expansion stroke, would not be fully utilized. In order to avert this, injector opening pressure was increased, which reduced fuel droplet size, increased surface to volume ratio and required comparatively less heat to evaporate vegetable oil droplet.

The trend exhibited by both versions of the combustion chamber with dual fuel operation at higher injector opening pressure of 270 bar was similar to the corresponding injector opening pressure of 190 bar. However, the maximum induction of alcohol was 40% in conventional engine at an injector opening pressure of 270 bar against 35% at 190 bar, while maximum alcohol induction remained same with engine with LHR combustion chamber at 270 bar as in the case of 190 bar.

From Table 4, it is noticed that, with pure diesel operation, engine with LHR combustion chamber increased the peak brake thermal efficiency by 3% at recommended injection timing and

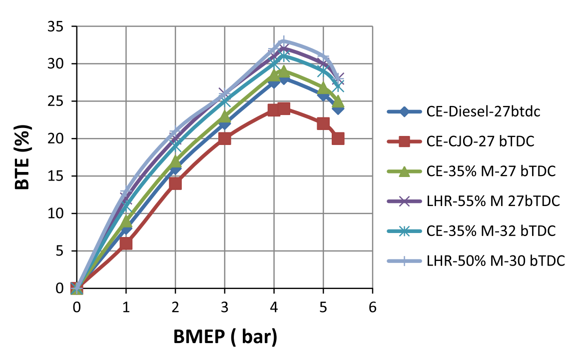


Fig. 5. Variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) with maximum percentage of methanol (M) induction in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

Table 4
Comparative data on peak brake thermal efficiency and brake specific energy consumption (BSEC) at full load operation.

IT/Fuel	Engine version	Methanol induction on mass basis (%)	Peak brake thermal efficiency (%)			BSEC (kWh) at full load operation		
			Injector opening pressure (bar)			Injector opening pressure (bar)		
			190	230	270	190	230	270
27/CJO	CE	0	24	25	26	4.9	4.7	4.65
		35	29	30	31	3.98	3.96	3.94
		40	–	–	32	–	–	3.88
	LHR	0	30	31	32	3.84	3.80	3.76
		55	32	33	34	3.76	3.74	3.72
30/CJO	LHR	0	31	32	33	3.72	3.68	3.66
		50	33	34	35	3.68	3.66	3.64
32/CJO	CE	0	28	29	30	4.0	3.96	3.92
		35	31	32	33	3.76	3.74	3.72
27/D	CE	0	28	29	30	4.0	3.92	3.84
		35	30	31	32	3.96	3.92	3.88
		40	–	–	33	–	–	3.84
	LHR	0	29	30	31	4.16	4.12	4.08
		50	33	34	35	3.80	3.76	3.72
29/D	LHR	0	30	31	32	3.96	3.92	3.88
		45	35	35.5	36	3.61	3.58	3.55
31/D	CE	0	31	31.5	32	3.76	3.72	3.68
		35	33	32.5	32	3.73	3.7	3.67

decreased by 3% at optimized injection timing in comparison with conventional engine (CE). This was due to improved combustion with improved oxygen–fuel ratios at recommended injection timing and reduction of ignition delay at optimum injection timing.

From the same table, it is evident that, with vegetable oil operation, peak brake thermal efficiency increased by 25% at recommended injection timing and 11% at optimized injection timing with engine with LHR combustion chamber in comparison with CE. This was due to improved combustion with improved heat release rate in engine with LHR combustion chamber [9].

With maximum induction of methanol, peak brake thermal efficiency increased by 10% at recommended injection timing and 6% at optimized injection timing with LHR combustion chamber in comparison with CE. This was due to higher amount of methanol substitution in the engine with LHR combustion chamber.

From Table 4, it is observed that peak brake thermal efficiency increased with increase of injector opening pressure and advanced injection timing in both versions of the combustion chamber with test fuels. This was due to improved spray characteristics of the fuel with increased injector opening pressure. This data was compared with diesel operation along with methanol induction, which showed higher peak brake thermal efficiency in comparison with crude jatropha oil operation with methanol induction. This was due to higher calorific value of diesel when compared with vegetable oil operation.

Conventional engine with crude jatropha oil operation gave higher brake specific energy consumption (BSEC) when compared with pure diesel operation. This was because of high viscosity, poor volatility and reduction in heating value of vegetable oil that lead to their poor atomization and combustion characteristics.

BSEC at full load operation decreased with advanced injection timing with test fuels. This was due to initiation of combustion at an early period.

From Table 4, it is noticed that, with pure diesel operation, engine with LHR combustion chamber increased brake specific energy consumption (BSEC) at full load operation by 4% at recommended injection timing and 5% at optimized injection timing in comparison with conventional engine (CE). As the combustion chamber was insulated to greater extent, it was expected that high combustion temperatures would be prevalent in LHR combustion chamber. It tends to decrease the ignition delay thereby reducing pre-mixed combustion as a result of which, less time was available for proper mixing of air and fuel in the combustion chamber leading to incomplete combustion, with which BSEC decreased at full load. Moreover at this load, friction and increased diffusion combustion resulted from reduced ignition delay. Increased radiation losses might have also contributed to the deterioration.

From the same table, it is evident that, with vegetable oil operation, BSEC at full load operation decreased by 22% at recommended injection timing and 7% at optimized injection timing with engine with LHR combustion chamber in comparison with CE. This was due to high viscosity, low volatility and low calorific value of the vegetable oil.

With maximum induction of methanol, BSEC at full load operation decreased by 5% at recommended injection timing and 2% at optimized injection timing with LHR combustion chamber in comparison with CE. This was due to higher amount of methanol substitution in the engine with LHR combustion chamber.

BSEC at full load operation decreased with the increase of methanol induction, as higher amount of methanol substitution caused improved evaporation and produced lower BSEC in both versions of the combustion chamber.

BSEC at full load operation decreased with the increase of injector opening pressure and advanced injection timing in both versions of the combustion chamber with test fuels. This was due to early initiation of combustion with improved fuel spray characteristics.

BSEC was lower with diesel operation along with carbureted methanol in comparison with crude jatropha oil operation with carbureted methanol. This was due to high calorific value of diesel.

Curves in Fig. 6 indicate that conventional engine with crude vegetable oil operation gave higher value of exhaust gas temperature when compared with other versions of the combustion chamber with other test fuels. Though the calorific value (or heat of combustion) of vegetable oil was less than that of diesel, density of the vegetable oil was higher; therefore greater amount of heat was released in the combustion chamber leading to higher exhaust gas temperature with conventional engine, which confirmed that performance deteriorated with conventional engine with vegetable oil operation in comparison with pure diesel operation.

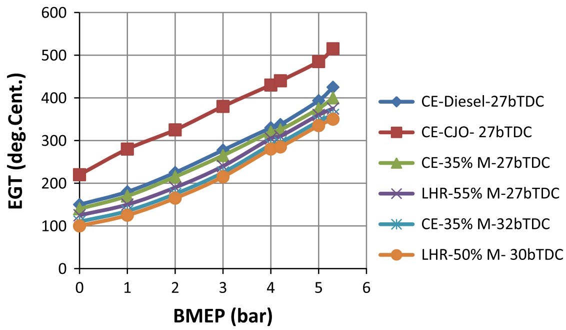


Fig. 6. Variation of exhaust gas temperature (EGT) with brake mean effective pressure (BMEP) with maximum percentage of methanol (M) induction in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

Similar findings were obtained by other studies [9]. This was also because of high duration of combustion of vegetable oil causing retarded heat release rate [9].

However, exhaust gas temperature decreased with the increase of percentage of methanol induction in both versions of the combustion chamber. At the recommended injection timing, the value of exhaust gas temperature was lower in conventional engine with 35% methanol induction at all loads when compared with pure diesel operation on conventional engine. Lower exhaust gas temperatures were observed in the engine with LHR combustion chamber with 55% methanol induction when compared with conventional engine with 35% methanol induction. This was due to higher amount of methanol substitution which absorbs combustion temperatures due to its high latent heat of evaporation. This showed that the performance of the engine with LHR combustion chamber improved with 55% methanol induction over conventional engine with 35% methanol induction.

Exhaust gas temperature further decreased, when the injection timings were advanced in both versions of the combustion chamber. This was because, when the injection timing was advanced, the work transfer from the piston to the gases in the cylinder at the end of the compression stroke was too large, leading to reduction in the value of exhaust gas temperature.

From Table 5, it is noticed that, with pure diesel operation, engine with LHR combustion chamber increased exhaust gas temperature at full load operation by 5% at recommended injection timing and 12% at optimized injection timing in comparison with conventional engine (CE). This indicated that heat rejection was restricted through the piston and liner, thus maintaining the hot combustion chamber as a result of which exhaust gas temperature increased with reduction of ignition delay [9].

From the same table, it is evident that, with vegetable oil operation, exhaust gas temperature at full load operation increased by 13% at recommended injection timing and 6% at optimized injection timing with engine with LHR combustion chamber in comparison with CE. This was due to high viscosity, low volatility and low calorific value of the vegetable oil.

With maximum induction of methanol, exhaust gas temperature at full load operation decreased by 6% at recommended injection timing and comparable at optimized injection timing with LHR combustion chamber in comparison with CE. This was due to higher amount of methanol substitution in the engine with LHR combustion chamber leading to reduction in the value of exhaust gas temperature due to its high latent heat of evaporation.

The value of exhaust gas temperature decreased marginally with increase of injector opening pressure in both versions of the combustion chamber as it is seen in Table 3. This is due to improved air–fuel ratios with increase of injection pressure. Exhaust gas temperatures were observed to be lower with diesel

Table 5
Comparative data on exhaust gas temperature (EGT) and coolant load at full load operation.

IT/Fuel	Engine version	Methanol induction on mass basis (%)	EGT at full load operation (°C) Injector opening pressure (bar)			Coolant load (CL) at full load operation (kW) Injector opening pressure (bar)		
			190	230	270	190	230	270
27/CJO	CE	0	515	490	480	4.4	4.6	4.8
		35	400	375	350	3.6	3.8	4.0
		40	–	–	320	–	–	0.3.8
	LHR	0	450	425	400	3.8	3.6	3.4
		55	375	350	325	3.4	3.2	3.0
30/CJO	LHR	0	430	410	390	3.6	3.4	3.2
		50	350	325	300	2.9	2.7	2.5
32/CJO	CE	0	460	430	400	4.6	4.8	5.0
		35	360	340	320	3.8	4.0	4.2
27/D	CE	0	425	410	395	4.0	4.2	4.4
		35	390	380	370	3.5	3.7	3.9
		40	–	–	360	–	–	3.7
	LHR	0	450	425	400	3.6	3.4	3.2
		50	360	350	340	3.3	3.2	3.1
29/D	LHR	0	420	400	380	3.2	3.0	2.8
		45	320	315	310	2.8	2.7	2.6
31/D	CE	0	375	350	325	4.2	4.4	4.6
		35	350	340	360	3.2	3.4	3.6

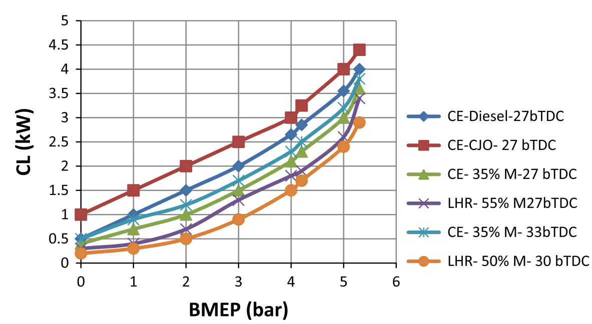


Fig. 7. Variation of coolant load (CL) with brake mean effective pressure (BMEP) with maximum percentage of methanol (M) induction in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

operation with carbureted methanol in comparison with crude jatropha oil operation with carbureted methanol. This was due to retarded ignition delay and higher duration of combustion with crude jatropha when compared with diesel [10].

From Fig. 7, it is seen that coolant load increased with increase of BMEP in both versions of the combustion chamber at recommended and optimized injection timings with methanol induction. This was due to increase of gas temperatures with load. It is observed that coolant load increased with crude jatropha oil operation, when compared with pure diesel operation. This was due to the increase of the un-burnt concentration at combustion chamber walls [9,10]. Coolant load was less in both versions of the combustion chamber at different percentages of methanol induction at all loads when compared with pure diesel operation on conventional engine. This was due to the reduction of gas temperatures with methanol induction. At recommended injection timing, cooling load was less in the engine with LHR combustion chamber with 55% methanol induction when compared with CE with 35% methanol induction at all loads. This was due to the thermal insulation provided in engine with LHR combustion chamber.

Cooling load increased in conventional engine, while it decreased in engine with LHR combustion chamber with the advanced injection timing. In case of conventional engine, unburnt fuel concentration reduced with effective utilization of energy, released from the combustion; coolant load with test fuels increased marginally at full load operation, with increase of gas temperatures, when the injection timing was advanced to the optimum value. However, the improvement in the performance of the conventional engine was due to heat addition at higher temperatures and rejection at lower temperatures, while the improvement in the efficiency of the engine with LHR combustion chamber was due to recovery from coolant load at their respective optimum injection timings with test fuels.

From Table 5, it is noticed that with pure diesel operation, coolant load at full load operation decreased by 10% at recommended injection timing and 24% at optimized injection timing with engine with LHR combustion chamber, in comparison with CE. This was due to the provision of thermal insulation with engine with LHR combustion chamber. This was also due to reduction of gas temperatures with improved oxygen–fuel ratios with LHR combustion chamber and increase of the same with conventional engine with pure diesel operation with advanced injection timing.

From Table 5, it is noticed that with vegetable oil operation, coolant load at full load operation decreased by 14% at recommended injection timing and 22% at optimized injection timing with engine with LHR combustion chamber, in comparison with CE [7]. This was due to improved combustion with provision of thermal insulation with engine with LHR combustion chamber.

With maximum induction of methanol, coolant load at full load operation is comparable at recommended injection timing and 29% at optimized injection timing, with engine with LHR combustion chamber in comparison with conventional engine. This was because of higher amount of methanol induction in engine with LHR combustion chamber leading to reduction in the exhaust gas temperature, due to its high latent heat of evaporation.

From Table 5, it is seen that coolant load increased marginally in CE, while it decreased in engine with LHR combustion chamber with increase of the injector opening pressure with test fuels. This was due to increase of injector opening pressure with conventional engine, increased nominal fuel spray velocity resulting in improved fuel–ir mixing with which gas temperatures increased. The reduction of coolant load in engine with LHR combustion chamber was not only due to the provision of the insulation but also it was due to improved fuel spray characteristics and increase of oxygen–fuel ratios causing decrease of gas temperatures and hence the coolant load.

Coolant load decreased marginally with increase of injector opening pressure in engine with LHR combustion chamber with test fuels. This was due to improved air–fuel ratios with which gas temperatures decreased and hence coolant load.

Cooling load with diesel operation along carbureted methanol was less in both versions of the engine in comparison with crude jatropha oil operation along with carbureted methanol. This was due to clean combustion with diesel operation.

Fig. 8 indicates that volumetric efficiency (VE) decrease with crude jatropha oil. This was due to increase of combustion chamber wall temperatures. Volumetric efficiency decreased marginally in both versions of the combustion chamber with methanol operation when compared with pure diesel operation on conventional engine; as percentage of methanol induction increased, the amount of air admitted into the cylinder of the engine reduced. However, conventional engine with different percentages of methanol induction showed higher volumetric efficiency when compared with engine with LHR combustion chamber. This was because of increase of temperature of insulated components in LHR combustion chamber, which heats the incoming charge to high temperatures and consequently the mass of air inducted in each cycle lowers.

Volumetric efficiency increased marginally with both versions of the combustion chamber with test fuels with advanced injection timing. This was due to decrease of combustion chamber wall temperatures with improved oxygen–fuel ratios.

From Table 6, it is observed that, with pure diesel operation, volumetric efficiency at full load operation decreased by 7% at recommended injection timing and 10% at optimized injection timing with engine with LHR combustion chamber, in comparison with CE. This was because of increase of temperatures of insulated components in LHR engine, which heats the incoming charge to high temperatures and consequently the mass of air inducted in each cycle lowers.

From the same table, it is clear that, with vegetable oil operation, volumetric efficiency at full load operation decreased by 4% at recommended injection timing and 8% at optimized

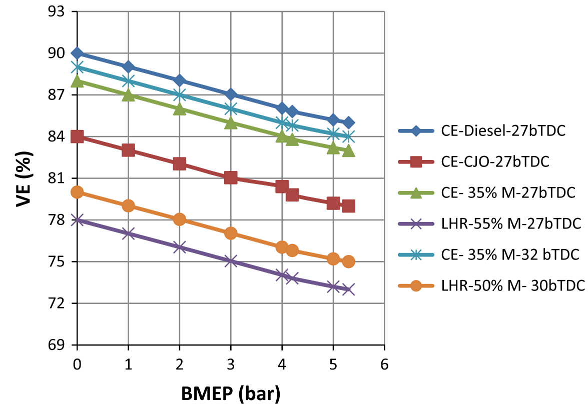


Fig. 8. Variation of volumetric efficiency (VE) with brake mean effective pressure (BMEP) with maximum percentage of methanol (M) induction in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

Table 6
Comparative data on volumetric efficiency and sound intensity at full load operation.

IT/Fuel	Engine version	Methanol induction on mass basis %	Volumetric efficiency (VE) at full load operation (%)			Sound intensity at full load operation (Decibels)		
			Injector opening pressure (bar)			Injector opening pressure (bar)		
			190	230	270	190	230	270
27/CJO	CE	0	81	82	83	95	90	85
		35	79	80	81	70	65	60
		40	–	–	82	–	–	50
	LHR	0	78	79	80	65	60	55
	LHR	55	73	75	77	60	55	50
30/CJO	LHR	0	79	80	81	60	55	50
		50	75	77	79	55	50	45
32/CJO	CE	0	86	87	88	75	70	65
		35	80	81	82	65	60	55
27/D	CE	0	85	86	87	85	80	75
		35	81	82	83	60	55	50
		40	–	–	82	–	–	45
	LHR	0	79	80	81	90	85	80
		50	78	79	80	50	45	40
29/D	LHR	0	80	81	82	75	70	65
		45	79	80	81	45	40	35
31/D	CE	0	89	90	91	60	55	50
		35	82	83	84	55	50	45

injection timing with engine with LHR combustion chamber, in comparison with CE.

With maximum induction of methanol, volumetric efficiency at full load operation decreased by 8% at recommended injection timing and 6% at optimized injection timing, with engine with LHR combustion chamber in comparison with conventional engine.

Volumetric efficiency increased marginally with both versions of the combustion chamber with test fuels with advanced injection timing. This was due to decrease of combustion chamber wall temperatures with improved oxygen–fuel ratios.

Volumetric efficiency increased marginally with the increase of injector opening pressure in both versions of the combustion chamber with test fuels. This was due to improvement of air utilization and combustion with the increase of injector opening pressure. However, these variations were very small.

Volumetric efficiency was marginally higher with diesel operation along with carbureted methanol compared with vegetable oil operation with carbureted methanol. This was due to increase of exhaust gas temperatures and hence combustion chamber wall temperatures.

If any fuel is to be tested as an alternate fuel, sound intensity is to be checked with alternate fuels with varied engine conditions. Fig. 9 indicates that sound levels increased with increase of brake mean effective pressure with test fuels. This was due to initiation of combustion with increase of load. Sound levels decreased marginally at 80% of the full load operation with test fuels with different versions of the combustion chamber due to efficient combustion with improved oxygen–fuel ratios. Sound intensities drastically increased in conventional engine with crude jatropha oil operation in comparison with conventional engine with pure diesel operation. This was due to deterioration in the performance of crude vegetable oil operation on conventional engine. High viscosity, poor volatility and high duration of combustion caused improper combustion of crude jatropha oil leading to generate high sound levels.

With higher amount of methanol induction, sound intensities were observed to be lower, due to efficient combustion in both versions of the combustion chamber. With methanol induction, engine with LHR combustion chamber gave lower levels of sound levels in comparison with conventional engine. This was due to increase of homogeneity of mixture with increased amount of methanol in engine with LHR combustion chamber.

When injection timings were advanced to optimum, sound intensities were reduced for both versions of the combustion chamber, due to early initiation of combustion.

From Table 6, it is observed that, with pure diesel operation, sound levels at full load operation decreased by 6% at recommended injection timing and 25% at optimized injection timing with engine with LHR combustion chamber, in comparison with conventional engine. Combustion deteriorated with diesel operation on engine with LHR combustion chamber due to reduction of ignition delay. This was also because of increase of sound velocity in adiabatic medium at high temperatures.

From the same table, it is clear that, with vegetable oil operation, sound levels at full load operation decreased by 31% at recommended injection timing and 20% at optimized injection timing with engine with LHR combustion chamber, in comparison with conventional engine. This was due to improved combustion in the hot environment provided by the engine with LHR combustion chamber.

With maximum induction of methanol, sound levels at full load operation decreased by 14% at recommended injection timing and 15% at optimized injection timing, with engine with LHR combustion chamber in comparison with conventional engine. This was due to higher amount of methanol induction leading to reduced combustion temperatures and improved combustion in LHR combustion chamber.

Table 6 denotes that the sound intensity decreased with increase of injector opening pressure for both versions of the combustion chamber with the test fuels. This was due to improved spray characteristic of the fuel, with which there was no impingement of the fuel on the walls of the combustion chamber leading to produce efficient combustion.

Sound intensities were observed to be lower with diesel operation along with carbureted methanol in comparison with CJO operation along with carbureted methanol. This was due to efficient combustion of diesel as it is less viscous and has a low duration of combustion.

3.2. Exhaust emissions

Fig. 10 indicates that during the first part, the particulate emissions were more or less constant, as there was always excess air present. However, in the higher load range there was an abrupt rise in particulate emissions due to less available oxygen, causing the decrease of oxygen–fuel ratio, leading to incomplete combustion, producing more smoke levels with test fuels. The variation of particulate emissions with the brake mean effective pressure typically showed an inverted L-shaped behavior due to the predominance of hydrocarbons in their composition at light load and of carbon at high load.

Particulate emissions drastically increased with crude jatropha oil operation with increase of BMEP. The increase of particulate emissions was also due to decrease of oxygen–fuel ratios and volumetric efficiency with crude vegetable oil operation when compared with pure diesel operation. Particulate emissions are related to the density of the fuel. Since vegetable oils have higher

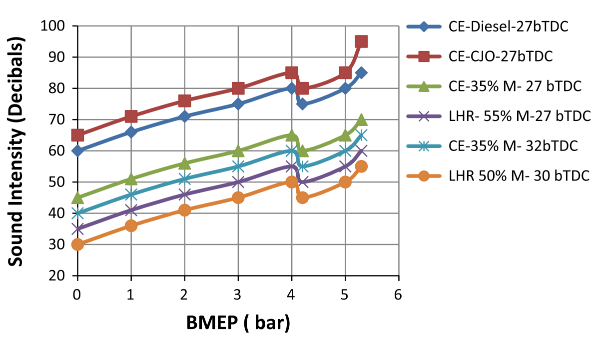


Fig. 9. Variation of sound intensity with brake mean effective pressure (BMEP) in conventional engine (CE) and engine with LHR combustion chamber at recommend injection timing and optimized injection timings at an injector opening pressure of 190 bar.

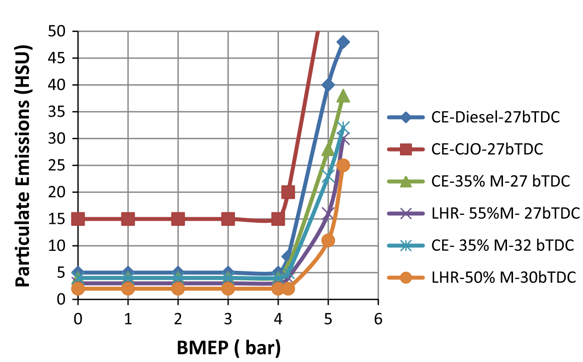


Fig. 10. Variation of particulate emissions in Hartridge smoke unit (HSU) with brake mean effective pressure (BMEP) with maximum percentage of methanol (M) induction in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

density compared to diesel fuels, particulate emissions were higher with vegetable oil operation.

However, the particulate emissions decreased with induction of methanol. The combustion of injected fuel in case of pure vegetable oil operation is predominantly one of oxidation of products of destructive decomposition. In this case, there are greater chances of fuel cracking and forming carbon particles. On the other hand, the combustion of alcohol is predominantly a process of hydroxylation and the chances of fuel cracking are negligible. Methanol does not contain carbon–carbon bonds and therefore cannot form any un-oxidized carbon particles or precursor to soot particles. One of the promising factors for reducing particulate emission with methanol was it contained oxygen in their composition which helped to reduce smoke levels.

Particulate emissions increased linearly with the increase of carbon to hydrogen atoms (C/H) ratio provided the equivalence ratio is not altered. This is because higher C/H lead to more concentration of carbon dioxide, which would be further reduced to carbon. Consequently, induction of methanol reduced the quantity of carbon particles in the exhaust gases as the values of C/H for diesel fuel, vegetable oil and methanol are 0.45, 0.7 and 0.25 respectively.

Lower particulate emissions were observed in both versions of the combustion chamber in dual fuel mode when compared with pure diesel operation on conventional engine. Engine with LHR combustion chamber with 55% methanol induction showed lower particulate emissions when compared with conventional engine with 35% methanol induction. Particulate emissions decreased with the increase of methanol induction in both versions of the engine. This was due to the reduction of gas temperatures as methanol has a high latent heat of evaporation. In dual fuel operation, particulate emissions further decreased with the advancement of the injection timing and with increase of injector opening pressure in both versions of the combustion chamber as it is noticed in Table 5, due to efficient combustion at higher injector opening pressure, which improved the atomization hence faster rate of combustion and shorter combustion duration at the advanced injection timings causing the reduction of the particulate emissions in both versions of the combustion chamber.

Particulate emissions were observed to be lower with diesel operation with carbureted methanol in comparison with crude vegetable oil operation with carbureted methanol. This was due to lower value of C/H ratio of diesel in comparison with vegetable oil which has got C/H value of 0.7.

From Table 7, it is observed that, with pure diesel operation, particulate emissions at full load operation increased by 25% at recommended injection timing and 42% at optimized injection timing with engine with LHR combustion chamber, in comparison with conventional engine. This was due to fuel cracking at higher temperature, leading to increase in smoke levels. Higher temperature of LHR combustion chamber produced increased rates of both particulate emissions and burn up. The reductions in volumetric efficiency and air–fuel ratio were responsible factors for increasing particulate emissions in the LHR combustion chamber at full load operation of the engine. As expected, particulate emissions increased in the LHR combustion chamber because of higher temperatures and improper utilization of the fuel consequent upon predominant diffusion combustion.

From the same table, it is clear that, with vegetable oil operation, particulate emissions at full load operation decreased by 19% at recommended injection timing and 18% at optimized injection timing with engine with LHR combustion chamber, in comparison with conventional engine. This was due to improved combustion in the hot environment provided by the engine with LHR combustion chamber.

With maximum induction of methanol, particulate emissions at full load operation decreased by 21% at recommended injection timing and 21% at optimized injection timing, with engine with LHR combustion chamber in comparison with conventional engine. This was due to higher amount of methanol induction in engine with LHR combustion chamber which caused improved combustion with reduction of fuel cracking temperatures.

Table 7
Comparative data on particulate emissions and NO_x levels at full load operation.

IT/Fuel	Engine version	Methanol induction on mass basis %	Particulate emissions at full load operation (HSU)			NO_x levels at full load operation (ppm)		
			Injector opening pressure (bar)			Injector opening pressure (bar)		
			190	230	270	190	230	270
27/CJO	CE	0	68	63	58	700	750	800
		35	38	33	28	425	475	525
		40	–	–	25	–	–	475
	LHR	0	55	50	45	1200	1150	1100
		55	30	27	25	650	600	550
30/CJO	LHR	0	45	40	35	1150	1100	1050
		50	25	23	21	400	350	300
32/CJO	CE	0	55	50	45	900	950	1000
		35	32	28	24	595	645	695
27/D	CE	0	48	38	34	850	900	950
		35	38	35	30	450	500	550
		40	–	–	28	–	–	500
	LHR	0	60	55	50	1150	1100	1050
		50	24	23	22	700	650	600
29/D	LHR	0	50	45	40	1125	1075	1025
		45	22	21	20	485	460	415
31/D	CE	0	35	30	25	1100	1150	1200
		35	29	28	27	650	700	750

Particulate emissions were observed to be lower with diesel operation along with carbureted methanol in comparison with CJO operation along with carbureted methanol. This was due to improved combustion with high cetane, less dense value of diesel with low value of C/H.

The temperature and the availability of oxygen are the reasons for the formation of NO_x. levels. It is noticed from Fig. 11, for both versions of the combustion chamber, NO_x levels rose steadily as the fuel/air ratio increased with increasing BMEP at constant injection timing with test fuels. At part load, NO_x levels were less in both versions of the combustion chamber. This was due to the availability of excess oxygen. At remaining loads, NO_x levels steadily increased with the load in both versions of the combustion chamber. This was because, local NO_x levels rose from the residual gas value following the start of combustion, to a peak at the point where the local burned gas equivalence ratio changed from lean to rich. At full load, with higher peak pressures, and hence temperatures, and larger regions close to stochiometric burned gas, NO_x levels increased in both versions of the combustion chamber.

NO_x levels were lower in conventional engine with vegetable oil operation when compared with diesel operation at all loads. This was due to lower heat release rate because of high duration of combustion causing lower gas temperatures with the vegetable oil operation on conventional engine which reduced NO_x levels [8].

However, NO_x emissions decreased with the increase of percentage of methanol induction in both versions of the combustion chamber, due to lower combustion temperatures. The low value of C/H ratio in methanol has indirect effect in reducing oxygen availability in the gases, which leads to the reduction of NO_x levels [23].

However, LHR combustion chamber with different percentages of methanol induction showed higher NO_x levels compared with CE with 35% methanol induction, due to increase of gas temperatures in LHR combustion chamber.

NO_x levels increased drastically in conventional engine while they decreased in LHR combustion chamber with the advancement of the

injection timing. This is due to reduction of gas temperatures in the LHR combustion chamber at its optimum injection timing. Increase of residence time and gas temperatures with conventional engine increased NO_x levels with both versions of the combustion chamber.

From Table 8, it is observed that, with pure diesel operation, NO_x emissions at full load operation increased by 36% at recommended injection timing and comparable at optimized injection timing with engine with LHR combustion chamber, in comparison with conventional engine. Due to the reduction in fuel–air equivalence ratio with LHR combustion chamber, which was approaching the stoichiometric ratio, NO_x levels increased.

From the same table, it is clear that, with vegetable oil operation, NO_x emissions at full load operation increased by 71% at recommended injection timing and 27% at optimized injection timing with engine with LHR combustion chamber, in comparison with conventional engine. This was due to increase of combustion temperatures in the hot environment provided by the engine with LHR combustion chamber.

With maximum induction of methanol, NO_x emissions at full load operation increased by 52% at recommended injection timing and 33% at optimized injection timing, with engine with LHR combustion chamber in comparison with conventional engine [24]. This was due to higher temperatures generated in the LHR combustion chamber.

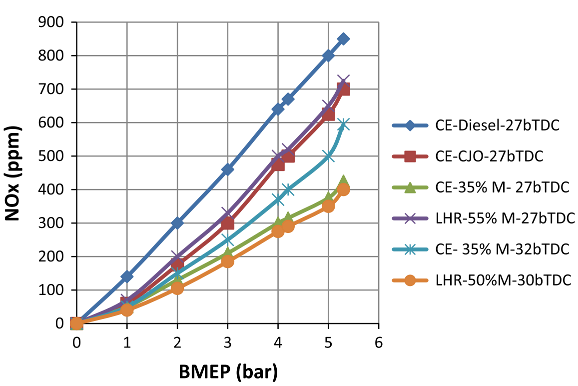


Fig. 11. Variation of NO_x levels with brake mean effective pressure (BMEP) with maximum percentage of methanol (M) induction in conventional engine (CE) and engine with LHR combustion chamber at recommended and optimum injection timings at an injector opening pressure of 190 bar.

NO_x levels decreased with increase of injector opening pressure in engine with LHR combustion chamber while they increased in conventional engine. This was due to decrease of gas temperatures with engine with LHR combustion chamber with improved air–fuel ratios and increase of the same with increase of injector opening pressure.

NO_x levels were observed to be higher with diesel operation along with carbureted methanol in comparison with CJO operation along with carbureted methanol in both versions of the engine. This was due to increase of gas temperatures with diesel operation as diesel has a higher calorific value than vegetable oil.

These aldehydes are responsible for pungent smell of the engine and affect human beings when inhaled in large quantities. The volatile aldehydes are eye and respiratory tract irritants. Though Government legislation has not been pronounced regarding the control of aldehyde emissions, when more and more alcohol engines are coming to existence severe measures for controlling aldehydes emitted out through the exhaust of the alcohol run engines will have to be taken up seriously.

It is observed in Table 9 that formaldehyde emissions were low with pure diesel operation in CE. Formaldehyde emissions increased drastically with methanol induction in both CE and LHR combustion chamber. With increased induction of methanol up to 35%, conventional engine registered very high value of formaldehyde emissions in the exhaust. Formaldehyde emissions decreased by 10% with conventional engine and 30% with LHR combustion chamber, when the injection timing was advanced to their optimum values, which showed the significant reduction in LHR combustion chamber. Hot environment of LHR engine completed combustion reactions and reduced the emissions of intermediate compounds, aldehydes. Hence it is concluded that LHR engine was more suitable for alcohol engines in comparison with pure diesel operation. Advanced injection timing and increase of injector opening pressure also improved the combustion performance in LHR combustion chamber by reducing the intermediate compounds like formaldehydes.

The trend of acetaldehyde emissions was similar to that of formaldehyde emissions as it is observed in Table 9. Both formaldehyde and acetaldehyde concentrations were observed to be lower with diesel operation along with carbureted methanol in both versions of the engine in comparison with CJO operation with carbureted methanol. This was due to clean combustion with diesel fuel and lower amount of intermediate compounds with diesel operation.

Table 8
Comparative data on formaldehyde and acetaldehyde concentrations at full load operation.

IT/Fuel	Engine version	Methanol induction on mass basis (%)	Formaldehyde concentration at full load operation (%) Injector opening pressure (bar)			Acetaldehyde concentration at full load operation () Injector opening pressure (bar)		
			190	230	270	190	230	270
27/CJO	CE	0	9.0	8.1	6.9	7	6	4.9
		35	28.3	26.2	24.1	18.3	16.4	14.7
		40	–	–	26.4	–	–	16.5
	LHR	0	7.0	6.8	6.6	5.0	4.6	4.4
		55	33.2	31.2	29.6	27.3	25.7	23.5
31/CJO	LHR	50	23.2	21.2	19.4	16	14.0.4	12.4
32/CJO	CE	0	6.2	5.8	5.4	5.8	5.6	5.4
		35	25.5	23.3	21.5	15.5	13.7	11.5
27/D	CE	0	6.0	5.5	5.0	5	4.5	4
		35	23.3	21.7	19.80	13	12.1	11.31
		40	–	–	22.13			12.61
	LHR	50	26.6	26.0	25.4	15.08	14.8	14.43
29/D	LHR	45	16.3	15.9	15.14	9.1	8.8	8.45
31/D	CE	0	5.2	5.0	4.8	4.6	4.4	4.2
		35	32.62	31.5	29.12	17.55	17.2	16.9

Table 9
Comparative data on peak pressure (PP), time of occurrence of peak pressure (TOPP) and maximum rate of pressure rise (MRPR) at full load operation.

IT/Fuel	Engine version	Methanol induction on mass basis (%)	PP at full load operation (bar) Injector opening pressure (bar)			TOPP at full load (deg) Injector opening pressure (bar)			MRPR at peak load operation (bar/deg) Injector opening pressure (bar)		
			190	230	270	190	230	270	190	230	270
27/CJO	CE	0	46.5	47.5	48.5	11	11	11	4.2	4.4	4.6
		35	52.6	53.7	54.8	8	8	7	5.6	5.8	6.0
		40	–	–	56.4	–	–	7			6.2
	LHR	0	58.6	56.6	54.6	7	7	6	5.8	5.6	5.4
		55	72.2	70.2	68.4	7	7	6	6.8	6.6	6.4
30/CJO	LHR	0	56.6	54.6	52.6	6	6	6	5.2	5.1	5.0
		50	68.4	66.4	64.4	6	6	6	6.0	5.8	5.6
32/CJO	CE	0	52.2	54.2	56.2	8	8	8	5.8	6.0	6.2
		35	60.2	61.2	62.2	7	7	7	6.5	6.7	6.9
27/D	CE	0	50.4	51.7	53.5	9	9	8	5.4	5.6	6.0
		35	53.6	54.6	56.0	8	7	7	4.4	4.6	4.8
		40	–	–	66.4	–	–	6	–	–	5.2
	LHR	0	64.6	62.6	60.6	6	6	6	5.2	5.4	5.6
	LHR	50	71.2	69.2	67.2	6	6	6	5.6	6.0	6.2
29/D	LHR	0	60.6	58.2	56.2	6	6	6	5.0	4.9	4.8
		45	69.2	67.2	65.2	7	6	6	6.0	6.2	6.4
31/D	CE	0	62.2	62.6	63.2	8	8	8	5.8	6.0	6.2
		35	63.2	63.8	64.2	8	7	7	6.2	6.4	6.6

3.3. Combustion characteristics

Crude jatropha oil with conventional engine at recommended injection timing gave lower value of peak pressure when compared with other versions of the combustion chamber with test fuels. This was due to low calorific value of the fuel with high duration of combustion with retarded heat release rates [8].

The value of peak pressure increased with advancement of the injection timing in conventional engine, while it decreased in the engine with LHR combustion chamber with test fuels. This was due to accumulated and sudden explosion of the fuel with advanced injection timing in conventional engine. Improved combustion with improved oxygen–fuel ratios resulted lower peak pressure with engine with LHR combustion chamber. This was also due to reduction of methanol induction at optimum injection timing in comparison with recommended injection timing.

From Table 9, it is observed that, with pure diesel operation, LHR version of the combustion chamber increased peak pressure by 20% at recommended injection timing and comparable at optimized injection timing in comparison with CE. This was because the LHR combustion chamber exhibited higher temperatures of combustion chamber walls leading to continuation of combustion, giving higher peak pressures. However, this phenomenon was nullified with an injection timing of 29° bTDC on the same LHR combustion chamber because of reduced temperature of combustion chamber walls thus bringing the peak pressures closer to TDC.

From the same table, it is noticed that, with vegetable oil operation, LHR version of the combustion chamber gave higher value of peak pressure by 26% at recommended injection timing and 8% at optimized injection timing in comparison with CE [7]. This was due to improved combustion of vegetable oil with engine with LHR combustion.

With maximum induction of methanol, engine with LHR combustion chamber gave higher value of peak pressure by 38% at recommended injection timing and 13% at optimized injection timing in comparison with CE. This was due to higher amount of substitution (60%) of methanol in engine with LHR combustion chamber at recommended injection timing and improved combustion at optimized injection timing [24].

Peak pressures increased with conventional engine with the increase of injector opening pressure with the test fuels. This may be due to smaller sauter mean diameter, shorter breakup length, improved dispersion and improved spray and atomization characteristics with conventional engine. This improves combustion rate in the premixed combustion phase. Peak pressure decreased with engine with LHR combustion chamber with test fuels with increase of injector opening pressure due to decrease of gas temperature with improved combustion.

Peak pressures were observed to be lower with vegetable oil operation with carbureted methanol in both versions of the engine in comparison with diesel operation with carbureted methanol. This was due to high calorific value of diesel fuel with high cetane number.

From the same table, it is observed that the value of time of occurrence (TOPP) decreased with the increase of methanol induction in both versions of the combustion chamber. When the methanol induction is increased to 55% in LHR combustion chamber, the value of TOPP was lower (shifted towards TDC) when compared with CE with 35% methanol induction.

This was once again confirmed by the observation of higher peak pressure and lower TOPP in engine with LHR combustion chamber, with dual fuel mode, that the performance of the engine with LHR combustion chamber with 55% methanol induction improved over conventional engine with 35% methanol induction. The value of TOPP decreased with the advancement of the injection timing with both versions of the combustion chamber. TOPP decreased with increase of injector opening pressure in both versions of the combustion chamber with methanol induction. This was due to improved air–fuel ratios with improved spray characteristics. TOPP was observed to be lower with diesel operation with carbureted methanol in comparison with vegetable oil operation with carbureted methanol. This was due to lower rate of combustion duration with low viscosity of diesel in comparison with vegetable oil.

From Table 9, it is evident that maximum rate of pressure rise (MRPR) was the highest for diesel operation, as diesel has high calorific value. MRPR increased with increase of methanol induction in both versions of the combustion chamber.

From Table 9, it is understood that, engine with LHR combustion chamber with 55% methanol induction at the recommended injection timing gave higher value of maximum rate of pressure rise (MRPR) in comparison with other versions of the combustion chamber with other test fuels. This was due to improved evaporation rate of methanol with hot insulated components of LHR combustion chamber.

Conventional engine with vegetable oil operation at recommended injection timing gave lower value of MRPR in comparison with other configurations of combustion chamber with other test fuels. This was due to retarded heat release rate and high duration of combustion of vegetable oil.

MRPR increased marginally with advanced injection timing with test fuels with conventional engine. This was because of initiation of combustion at early period and causing sudden explosion of fuel–air mixture. However, MRPR decreased marginally with advanced injection timing with engine with LHR combustion chamber with test fuels. This was due to decrease of combustion temperatures in engine with LHR combustion chamber with advanced injection timing.

From Table 9, it is observed that, with pure diesel operation, LHR version of the combustion chamber gave comparable value of MRPR at full load at recommended injection timing and increased by 13% at optimized injection timing in comparison with CE. This was due to reduction of ignition delay with diesel.

From the same table, it is noticed that, with vegetable oil operation, LHR version of the combustion chamber gave higher value of MRPR by 38% at recommended injection timing and 10% at optimized injection timing in comparison with CE. This was due to improved combustion of vegetable oil with engine with LHR combustion chamber.

With maximum induction of methanol, engine with LHR combustion chamber gave higher value of MRPR by 21% at recommended injection timing and 8% at optimized injection timing in comparison with conventional engine. This was due to higher amount of substitution (60%) of methanol in engine with LHR combustion chamber at recommended injection timing and improved combustion at optimized injection timing.

MRPR increased marginally with CE and decreased the same with engine with LHR combustion chamber with test fuels with increase of injector opening pressure at recommended and optimized injection timings. This was due to improved spray characteristics of the fuel. MRPR was highest for dual fuel operation followed by pure diesel operation on both versions of the combustion chamber.

4. Summary

1. The optimum injection timing was 32° bTDC with conventional engine, while it was 29° bTDC for engine with LHR combustion chamber with vegetable oil operation.
2. With conventional engine, at recommended injection timing of 27° bTDC, the performance with vegetable oil operation deteriorated, while it was comparable at optimum injection timing in comparison with pure diesel operation.
3. With vegetable oil operation, at recommended and optimized injection timings, the performance of the engine with LHR combustion chamber improved when compared with conventional engine.
4. The maximum induction of methanol was found to be 35% with conventional engine, while it was 55% with engine with LHR combustion chamber at recommended injection timing on mass basis of vegetable oil at full load operation.
5. The maximum induction of methanol was observed to be 35% with conventional engine, while it was 50% with engine with LHR combustion chamber at optimized injection timing on mass basis of vegetable oil at full load operation.
6. With maximum induction of methanol, at an injector opening pressure of 190 bar, at its optimized injection timing, engine with LHR combustion chamber increased peak brake thermal efficiency by 6%, at full load operation – brake specific energy consumption was comparable, decreased exhaust gas temperature by 19%, coolant load by 19%, volumetric efficiency by 5%, sound levels by 8%, particulate matter by 44%, NO_x emissions by 65%, increased formaldehyde emissions and acetaldehyde emissions drastically, increased peak pressure by 21% and maximum rate of pressure rise by 15% when compared with crude jatropha oil operation at similar operating conditions.

4.1. Future scope of the work

1. Investigations can also be carried out to evaluate the performance of indirect injection diesel engine with test fuels.
2. Studies can be made with other versions of the LHR combustion chamber with carbureted alcohol (ethanol/methanol) and wide variety of non-edible vegetable oil.
3. A suitable catalytic converter can be designed to control formaldehyde and acetaldehyde emissions from engine run with alcohol with less expensive and easily available catalysts.

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